

Development of a new solar thermal engine system for circulating water for aeration

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Abstract

This paper presents a numerical study about the performance of a Beta Stirling solar thermal engine system. This system is composed of a solar collector box connected to a regenerator hydraulic system and a transmitting power system. The objective of the system is to offer a new alternative to help solving stagnant water pollution in hot countries like Thailand by circulating water in canals, lakes, ponds etc. for aeration using solar energy.

The purpose of this study is to determine the power output and actual heat transfer on the performance of the solar thermal engine. The solar thermal engine is analyzed using a mathematical model based on the first law of thermodynamics for processes with finite speed, with particular attention to the energy balance at the receiver. The result of calculations showed that the regenerator volume and phase angle must be chosen carefully to fulfill the requirement that total fluid mass in the system is constant and to obtain maximum power output throughout the day.

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1. Introduction

Environmental impact and depletion of mineral resources such as gas oil and lignite are prompting a re-

examination of an alternative to these resources. Renewable energy sources seem to be a good alternative. The power needed to drive turbine to circulate water of canals, lakes ponds, etc. must come from local energy sources. Besides electrical motor power, solar and wind energy, and to some extent biogas, may provide the solution to this problem.

Solar energy is the most fascinating and promising renewable energy source. Thailand is a country situated within an equatorial belt having hot and humid climate. The average daily intensity of solar radiation is 17.5 MJ/

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Nomenclature

A_C	collector area (m^2)	T_{plate}	mean plate temperature (K)
dx	phase angle	T_L	intermediate low temperature sink (K)
n	engine speeds (rps)	U_L	coefficient of the thermal energy lost ($\text{W}/\text{m}^2\text{K}$)
P_{mean}	mean engine pressure (Pa)	V	volume (m^3)
P_{min}	minimum engine pressure (Pa)	V_{SE}	swept volume of expansion piston or displacer piston (m^3)
P_{max}	maximum engine pressure (Pa)	V_{SC}	swept volume of compression piston or power piston (m^3)
P_{W_i}	theoretical output engine power (W)	V_B	overlap volume (m^3)
Q_H	heat transfer (hot side) (kJ)	V_{DC}	compression dead volume (m^3)
Q_C	heat transfer (cold side) (kJ)	V_{DE}	dead volume of expansion space (m^3)
R	gas constant	V_R	regenerator volume (m^3)
S	entropy of the working fluid (kJ/kg K)	x	crankshaft angle (degree)
s	solar radiation absorbed by a collector (W/m^2)	W_C	energy of the compression (J)
t	temperature ratio of T_E/T_C (K/K)	W_E	energy of the expansion (J)
T_a	ambient temperature (K)	W_i	energy output (J)
T_C	constant cold temperature of the working fluid in the compression space (K)	η_e	thermal efficiency
T_E	constant hot temperature of the working fluid in the expansion space (K)		

m^2day (Namprakai et al., 1989). Therefore, it is necessary to consider the use of this freely available energy in various fields such as the conservation of food products, the habitat, and its transformation into mechanical energy. In recent decades, some solar pump and solar thermal engines operating on the principle of thermodynamic conversion scheme have been built and tested extensively throughout the world (Kaushik and Kumar, 2000). These systems used the thermal energy produced by the solar energy to drive a conventional engine for pumping water (Tallbert et al., 1978). However, many new designs have been introduced recently and the developments in this field need to be updated. The aim of this paper is to study the performance of a solar thermal system for circulating water in canals, lakes ponds etc. for aeration. In fact, water pollution, is not only due to garbage and waste disposed into the water, but also as a result of poor circulation of water or water stagnation in many canals, lakes and ponds. This water, usually in urban and suburban areas, is enclosed and has weak wind speed to help in the circulation.

2. Literature review

The objective of this review is to provide a basic background and some review of existing literature on solar-powered Stirling engine technology that uses low temperature technology. A number of Stirling engine configurations and designs, including the engine's development, are provided and discussed.

The extent of research on solar Stirling-cycle for power production has been limited to fraction of horsepower (Farber and Prescott, 1965). The hot-air engine developed by Farber and Prescott in 1965 focused on area where air was heated and its expansion pushes the piston down. In the down-stroke of the piston, the displacer moved to the left by a linkage. On the up-stroke of the piston, the displacer moved to the right and all the hot air was at the left section of the cylinder and lost heat to the cooling water. An efficiency of about 9% was obtained at 100 rpm with a break horsepower of about 0.2. And after that Beale et al. (Beale et al., 1971) in 1971, considered a free cylinder containing a heavy piston which remained essentially stationary and a light displacer which moved under the influence of pressure differential between the work space and the bound space in the cylinder. The cylinder moved under the influence of the same pressure differential and performs work against an external load. Since the entire pressure enclosure moved as a unit, the free cylinder engine can deliver work from a completely sealed working gas and since there are no internal bearing loads, or mechanical linkage or gears, etc., no lubrication was needed. The engine should be inexpensive and should have a long life. A Fresnel lens focused the solar radiation on the hot space of the engine and a double acting water pump utilized the developed mechanical energy. An efficiency of about 9% was obtained at 100 rpm with a power of about 149 W.

Stirling engine theoretically could achieve the highest possible energy conversion efficiency of all heat/power-engines. Using practically any kind of energy, they can

be employed with extremely low emissions as generator, motor, heat pump or cooling system. In spite of all these advantages, Stirling engines still have not made the market breakthrough as a mass product. A general analysis of finite time thermodynamics of a Stirling heat engine was presented with finite heat capacity of external reservoirs, regenerative losses and finite effectiveness of each of the heat exchangers (i.e. a high temperature heat exchanger and a low temperature heat exchanger) (Fette, 1995). They obtained the expressions for maximum power output and computed the corresponding thermal efficiency. The effect of operating temperatures, the effectiveness of the regenerative heat exchanger on the heat transfer (Q_H and Q_L) to and from the heat engine, the regenerative heat transfer (Q_R), the maximum power output (P) and the corresponding thermal efficiency of the cycle have all been studied. They considered the inlet source temperature and sink temperature as 1300 K and 300 K. They also used effectiveness of each heat exchanger and the regenerator in the range from 0.4 to 1, and the capacitance rates of the heat source/sink reservoirs in the range from 0.30 to 1.80 kW/K. Their analysis showed that power output and thermal efficiency were 62.55 kW and 43.94%, respectively.

3. Description of solar thermal engine

The solar thermal engine is essentially composed of a solar collector, a displacer, a cooling system, a regenerator, a hydraulic and a transmitting power system (Fig. 1).

The solar collector is a box, where the absorber is an aluminum plate, and the cover is a transparent acrylic

(2-layer fluor-acrylic) polymer. The thermal radiation absorbed by the absorber is used to heat the liquid ethyl fluorocarbons (R-11) and the absorbed energy set the fluid in motion by thermo-siphon action and the liquid is transformed into vapor. We are aware that R-11 is not environment friendly. However, due to its properties, which are suitable to operate the engine, we conduct our theoretical study using R-11 so that we could formulate a basis for the design of such a thermal engine. The outside of the box next to the regenerator is covered with insulation material to prevent heat loss to the outside.

The displacer moves up and down part of the box. When the displacer moves up it pushes the working vapor against the inside of the hot absorber plate. A displacer membrane, made of a flexible and heat-insulation fabric and attached to the outside ring of the displacer, connects the displacer to the interior lining on the outside of the solar collector box.

The cooling system located below the regenerator further cools down the temperature of the working fluid before it enters the lower and cool part of the solar collector box. The cooling of water-fed pipes absorbs heat from the working fluid flowing through. Cold water is circulated through the cooling system and thus keeps this area cool.

The hydraulic system is a heat exchanger between water and ethyl fluorocarbon vapor. The transmitting power system transforms the translation movement into a rotating movement. The function of the power piston is to convert the expansion of the working fluid at high pressure and to compress the working fluid at low temperature and to transfer this conversion into motion by means of a crankshaft and flywheel.

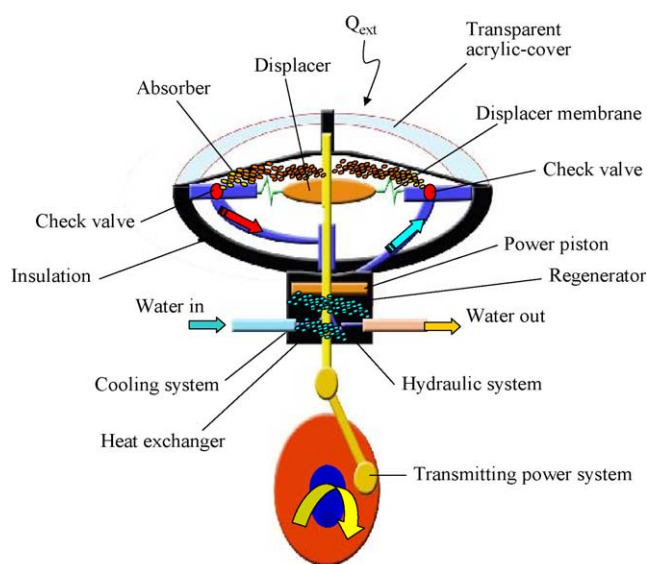


Fig. 1. Schematic diagram of the solar thermal engine.

When the displacer has reached the bottom position, cool vapor of ethyl fluorocarbons fills the box. So, the temperature and the pressure of the vapor reached its low value (about 20–30°C). When the displacer moves up, the process is reversed. Cool vapor of ethyl fluorocarbons is pushed through the regenerator picking up the heat stored there flowing along the inner hot side of the box, and its temperature increases. As soon as the displacer has reached its highest temperature (about 70–100°C) there is a fluctuation of the pressure in the box. The bottom of the box is attached to the absorber structure through a flexible membrane coupling system. Therefore, it can serve as a large-size power piston in response to the cyclical pressure fluctuation inside the box. The pressure fluctuation is used to generate mechanical work in the following way. The displacer moves the vapor cyclically from the hot to the cool section of the box. So, the evolution of the pressure flow is cyclical and subsequently the displacer movement produces a mechanical work. For half a cycle, the regenerator stores heat of the working fluid vapor temporarily and the cooling system is used to cool the vapor before it flows in the lower part of the box.

4. Theoretical model

4.1. The thermal engine

A simplified model (Hsu et al., 2003) as shown in Fig. 2 can represent the heat process of the solar thermal engine. The T – S diagram (Blank and Wu, 1995; Berrin Erbay and Yavuz, 1997) given in Fig. 3, shows the effects of heat transfer, regeneration time, and imperfect regeneration on the performance of the irreversible Stirling engine cycle.

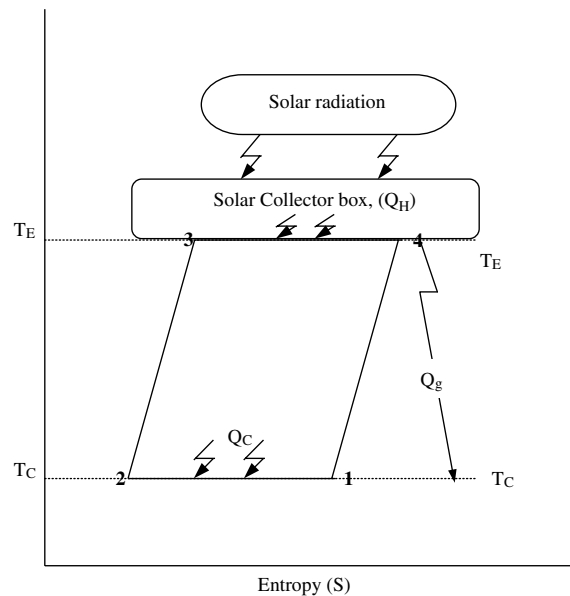


Fig. 3. T – S diagram of the solar thermal engine cycle.

The hypotheses for modeling the thermal engine are common. They are as follows:

- The expansion and the compression processes are isothermal.
- The vapor is considered as an ideal gas.
- There is a perfect regeneration.
- The closed system has constant fluid mass.
- The regenerator gas temperature is an average of the expansion gas temperature (T_E) and of the compression gas temperature (T_C).

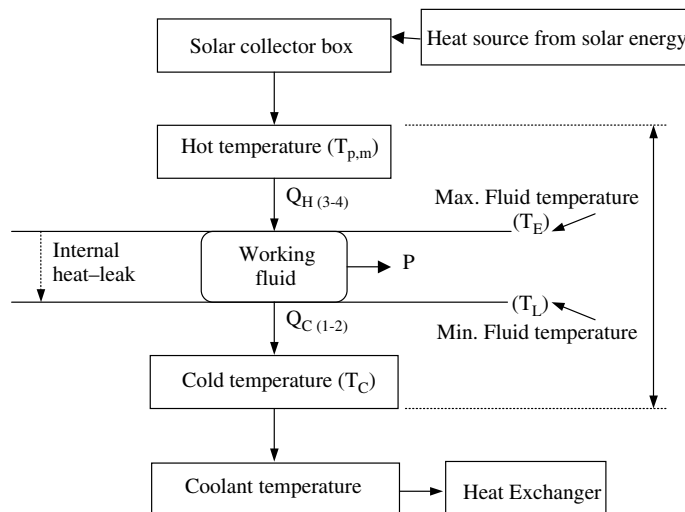


Fig. 2. The simplified heat transfer model of the solar thermal engine.

- The expansion space (V_E) and the compression space (V_C) change according to a sine curve.

The variation of the volume ratio in the engine system is performed by varying the volume of working liquid within the collector box below the working fluid vapor cushion.

The thermal efficiency of the process is defined as the ratio between “ W ” the external work output done per cycle and “ Q_{ext} ” the external heat supplied by solar energy per cycle to the engine (Costea and Feidt, 1998).

$$\eta = \frac{W}{Q_{\text{ext}}} \quad (1)$$

Concerning all phases the efficiency of the process of Fig. 2 is

$$\eta = \frac{Q_{3-4} - Q_{1-2}}{Q_{3-4} - (Q_{2-3} + Q_{4-1})} \quad (2)$$

In this ideal Stirling process, the regenerator efficiency is assumed to be equal to 100%, where Q_{ext} = external heat supplied by cycle (kJ); Q_{1-2} = heat rejected by the cooler at the compression phase at “ T_C ” (kJ); Q_{2-3} = isochoric heat supplied by V_{min} (kJ); Q_{3-4} = external heat supplied by the expansion phase at “ T_E ” (kJ); Q_{4-1} = isochoric heat rejected by V_{max} (kJ).

So $Q_{2-3} + Q_{4-1} = 0$ and the external supplied heat for the ideal process is $Q_{\text{ext}} = Q_{3-4}$. So that the ideal efficiency is

$$\eta_i = \frac{Q_{3-4} - Q_{1-2}}{Q_{3-4}} \quad (3)$$

Following the analysis conducted by Hirata (Hirata, 1997) on Beta Stirling engine, the momental expansion volume (V_E) and compression volume (V_C), described at a given crank angle, based on above assumptions are:

$$V_E = \frac{V_{\text{SE}}}{2} (1 - \cos(x)) + V_{\text{DE}} \quad (4)$$

$$V_C = \frac{V_{\text{SE}}}{2} (1 - \cos(x)) + \frac{V_{\text{SC}}}{2} \{1 - \cos(x - dx)\} + V_{\text{DC}} - V_B \quad (5)$$

where x = crankshaft angle (degree); V = volume (m^3); V_{SE} = swept volume of expansion piston or displacer piston (m^3); V_{SC} = swept volume of compression piston or power piston (m^3); V_B = overlap volume (m^3); V_{DC} = compression dead volume (m^3); V_{DE} = dead volume of expansion space (m^3); V_R = regenerator volume (m^3).

The overlap volume (V_B) is given by

$$V_B = \frac{V_{\text{SE}} + V_{\text{SC}}}{2} - \sqrt{\frac{V_{\text{SE}}^2 + V_{\text{SC}}^2}{4} - \frac{V_{\text{SE}} * V_{\text{SC}}}{2} \cos(dx)} \quad (6)$$

The volumes of the expansion and compression cylinder at a given crank angle are determined at first. The momental volumes are described with a crank angle (x). This crank angle is defined as $x = 0$ when the expansion piston is located at the top most position (top dead point).

The total momental volume (V) is the sum

$$V = V_E + V_C + V_R \quad (7)$$

The working fluid pressure (P) is deduced from the expression:

$$P = \frac{P_{\text{mean}} \sqrt{1 - C^2}}{1 - C \cdot \cos(x - a)} = \frac{P_{\text{min}} (1 + C)}{1 - C \cdot \cos(x - a)} = \frac{P_{\text{max}} (1 - C)}{1 - C \cdot \cos(x - a)} \quad (8)$$

The energy of the expansion (W_E) and compression (W_C) spaces are expressed respectively by

$$W_E = \oint P dV_E = \frac{P_{\text{mean}} V_{\text{SE}} \pi C \cdot \sin(a)}{1 + \sqrt{1 - C^2}} = \frac{P_{\text{min}} V_{\text{SE}} \pi \cdot C \cdot \sin(a)}{1 + \sqrt{1 - C^2}} * \frac{\sqrt{1 + C}}{\sqrt{1 - C}} = \frac{P_{\text{max}} V_{\text{SE}} \pi \cdot C \cdot \sin(a)}{1 + \sqrt{1 - C^2}} * \frac{\sqrt{1 - C}}{\sqrt{1 + C}} \quad (9)$$

$$W_C = - \oint P dV_C = - \frac{P_{\text{mean}} V_{\text{SE}} \cdot \pi \cdot C \cdot t \cdot \sin(a)}{1 + \sqrt{1 - C^2}} = - \frac{P_{\text{min}} V_{\text{SE}} \cdot \pi \cdot C \cdot t \cdot \sin(a)}{1 + \sqrt{1 - C^2}} * \frac{\sqrt{1 + C}}{\sqrt{1 - C}} = - \frac{P_{\text{max}} V_{\text{SE}} \cdot \pi \cdot C \cdot t \cdot \sin(a)}{1 + \sqrt{1 - C^2}} * \frac{\sqrt{1 - C}}{\sqrt{1 + C}} \quad (10)$$

$$C = \frac{B}{S} \quad (10a)$$

$$S = t + 2tX_{\text{DE}} + \frac{4tX_R}{1 + t} + V + 2X_{\text{DC}} + 1 \quad (10b)$$

$$B = \sqrt{t^2 + 2(t - 1)V \cos(dx) + V^2 - 2t + 1} \quad (10c)$$

$$a = \tan^{-1} \frac{V \cdot \sin(dx)}{t + \cos(dx) + 1} \quad (10d)$$

where P_{mean} = mean pressure (Pa); P_{min} = minimum pressure (Pa); P_{max} = maximum pressure (Pa); t = temperature ratio of T_E/T_C .

For one cycle, the energy is equal to

$$W_i = W_E + W_C \quad (11)$$

where W_E = energy of the expansion (J); W_C = energy of the compression (J); W_i = energy output (J).

The engine power (P_{W_i}) and the thermal efficiency (η_e) are defined respectively:

$$P_{W_i} = W_i n; \quad \eta_e = \frac{W_i}{W_E} = 1 - t \quad (12)$$

where n = engine speeds (rps).

4.2. The solar collector box

An energy balance that indicates the distribution of incident solar energy into useful energy gain, thermal losses, and optical losses describes the theoretical consideration of system of the solar collector box. The solar radiation absorbed by solar collector box is equal to the difference between the incident solar radiation and the optical losses. In steady state, the useful energy output of a collector is then the difference between the absorbed solar radiation and the thermal loss (Duffie and Beckman, 1991):

$$Q_u = A_C [S - U_L (T_{P,m} - T_a)] \quad (13)$$

where U_L = coefficient of the thermal energy lost (W/m^2); S = solar radiation absorbed by a collector (W/m^2); $T_{P,m}$ = mean plate temperature (K); T_a = ambient temperature (K); A_C = solar collector box area (m^2).

5. Results and discussion

To discuss the performance of this solar thermal engine, we use the data for the solar radiation and ambient temperature on the date 2/02/2001 in Bangkok, Thailand.

The parameters used in the simulation were

$$\begin{aligned} V_{SE} &= 0.0269 \text{ (m}^3\text{)}, \quad V_{SC} = 0.0269 \text{ (m}^3\text{)}, \\ V_{DC} &= 0.0237 \text{ (m}^3\text{)}, \quad V_{DE} = 0.0237 \text{ (m}^3\text{)}, \\ V_R &= 0.05 \text{ to } 0.5 \text{ (m}^3\text{)}, \\ T_E &= 50 + 273 \text{ (K) to } 200 + 273 \text{ (K)}, \\ T_C &= 30 + 273 \text{ (K)}, \quad P_{\text{mean}} = 180 \text{ (kPa)}, \\ A_C &= 0.39 \text{ (m}^2\text{)}. \end{aligned}$$

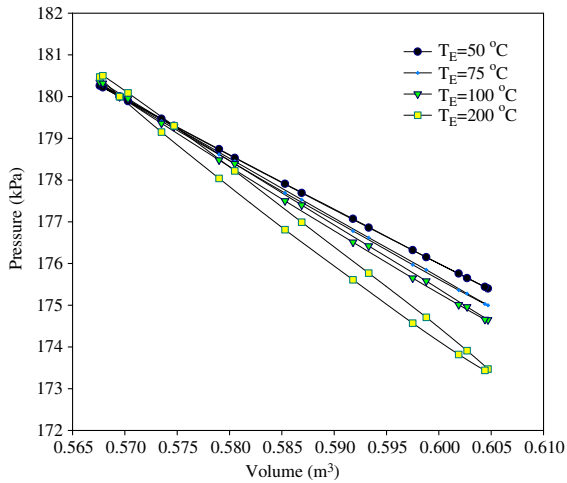


Fig. 4. Simulated P – V diagram for phase angle 90° at various temperature.

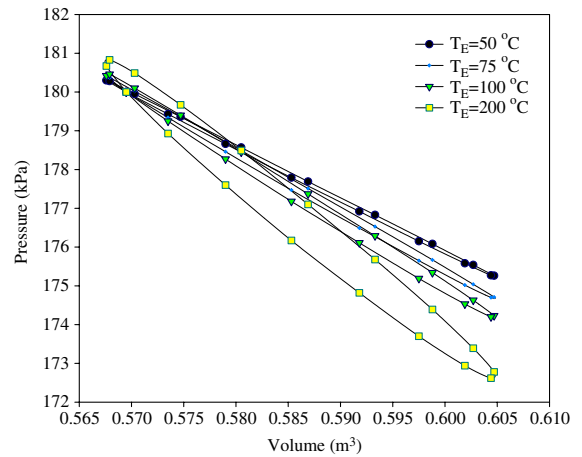


Fig. 5. Simulated P – V diagram for phase angle 140° at various temperature.

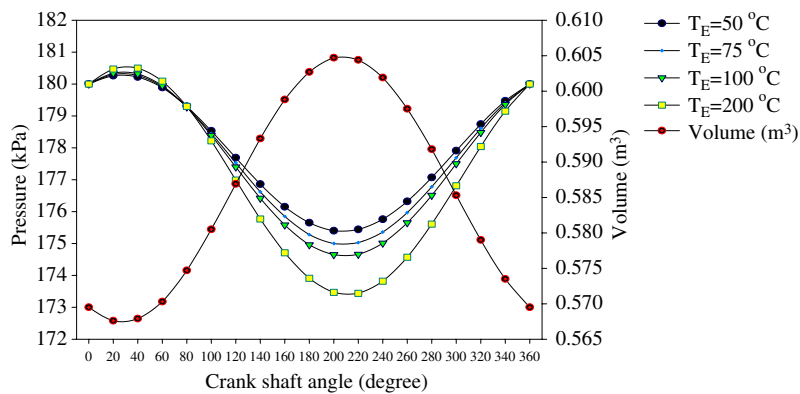


Fig. 6. Pressure and volume when phase angle is 90° ($V_R = 0.5 \text{ m}^3$).

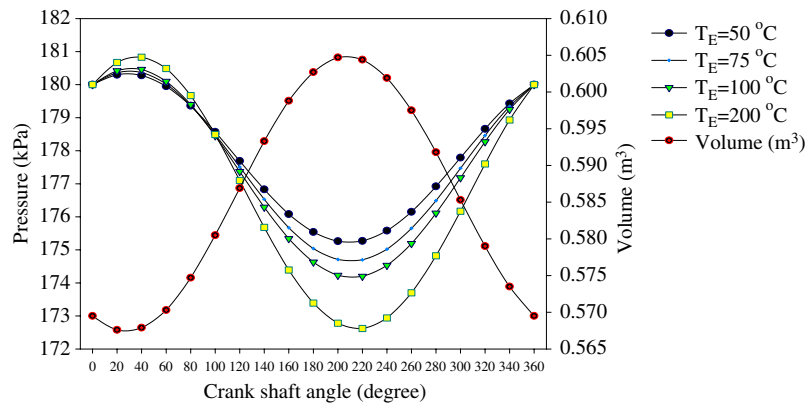


Fig. 7. Pressure and volume when phase angle is 140° ($V_R = 0.5 \text{ m}^3$).

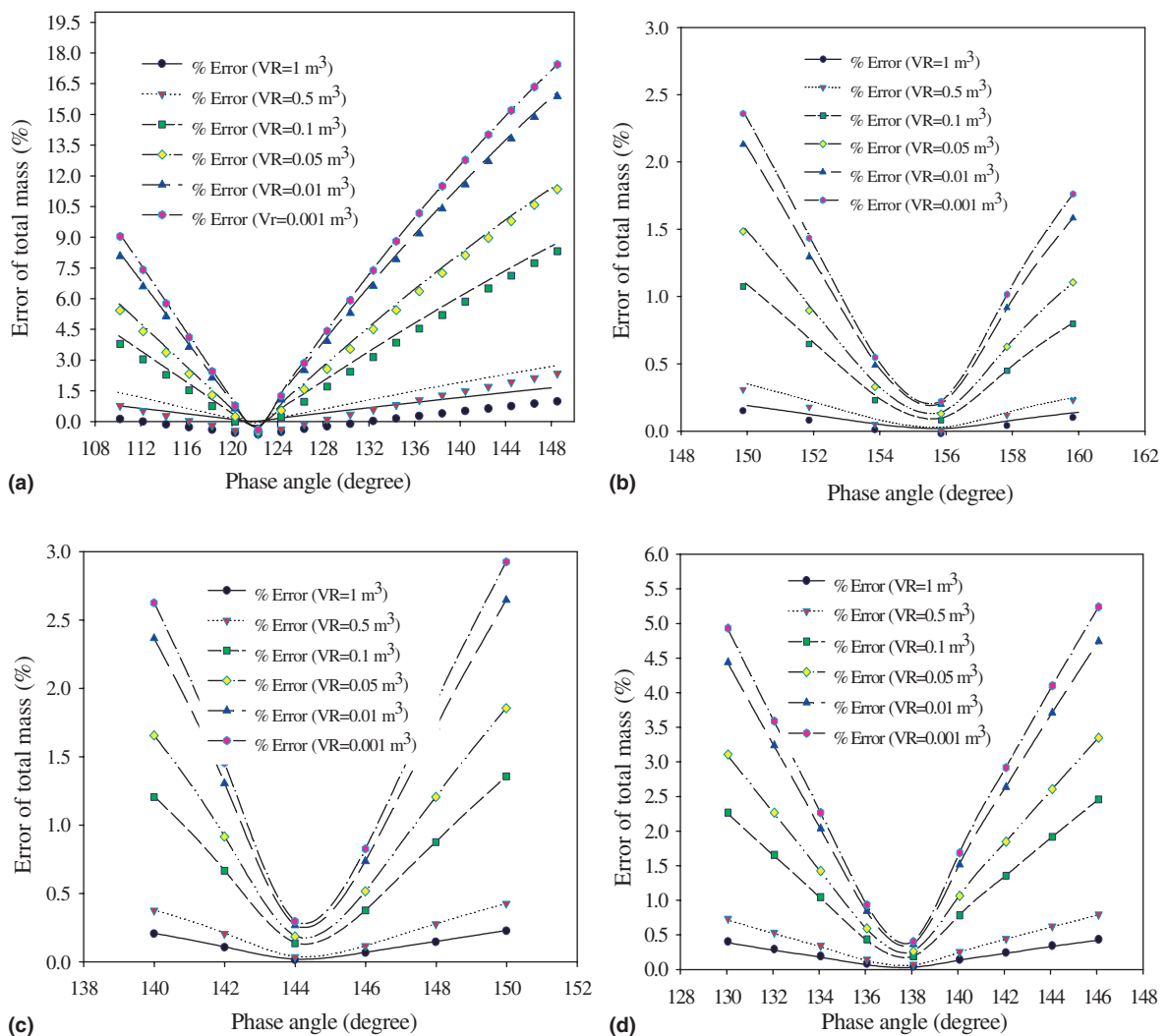


Fig. 8. Graphs of error of total mass at different phase angle and various T_E : (a) 200°C ; (b) 100°C ; (c) 75°C ; (d) 50°C .

Fig. 4 shows the simulated P – V diagram for 90° phase angle at various gas expansion temperature, and Fig. 5 shows that for 140° phase angle. Figs. 6 and 7 show the pressure variation versus the crankshaft angle at different gas expansion temperature. Obviously, the higher is the temperature, the higher the pressure and the larger the difference between the compression and expansion portions of the curves of the cycle; i.e., higher power work output.

One of the assumptions of Schmidt's theoretical model (Hirata, 1997) was that total mass of gases in the system was constant because it was a closed system. A check of this assumption by simulation showed that total mass would not be the same when the phase angle was varied, as shown in Fig. 8. The graphs show that the phase angle at which the minimum error in total mass varies with T_E (hot-side temperature) and independent of V_R . The values of the angles are shown in Table 1. However, V_R affects the slope of hot-side temperature Vs phase angle curve. When $V_R < 0.1 \text{ m}^3$, the slopes

Table 1

Phase angle at which error in total mass of vapor is minimum (1.25%)

Hot-side temperature ($^\circ\text{C}$)	Phase angle at which total mass is approximately constant
50	155
75	144
100	138
200	122

are steeper than when $V_R > 0.5 \text{ m}^3$. The maximum error was estimated to be 1.25%.

Since solar radiation varies throughout the day, the hot-side temperature can be expected to vary between 50°C and up to about 200°C . To accommodate this temperature variation, phase angle must be varied between 122° and 155° . It will be impractical to vary the phase angle according to the solar time through out

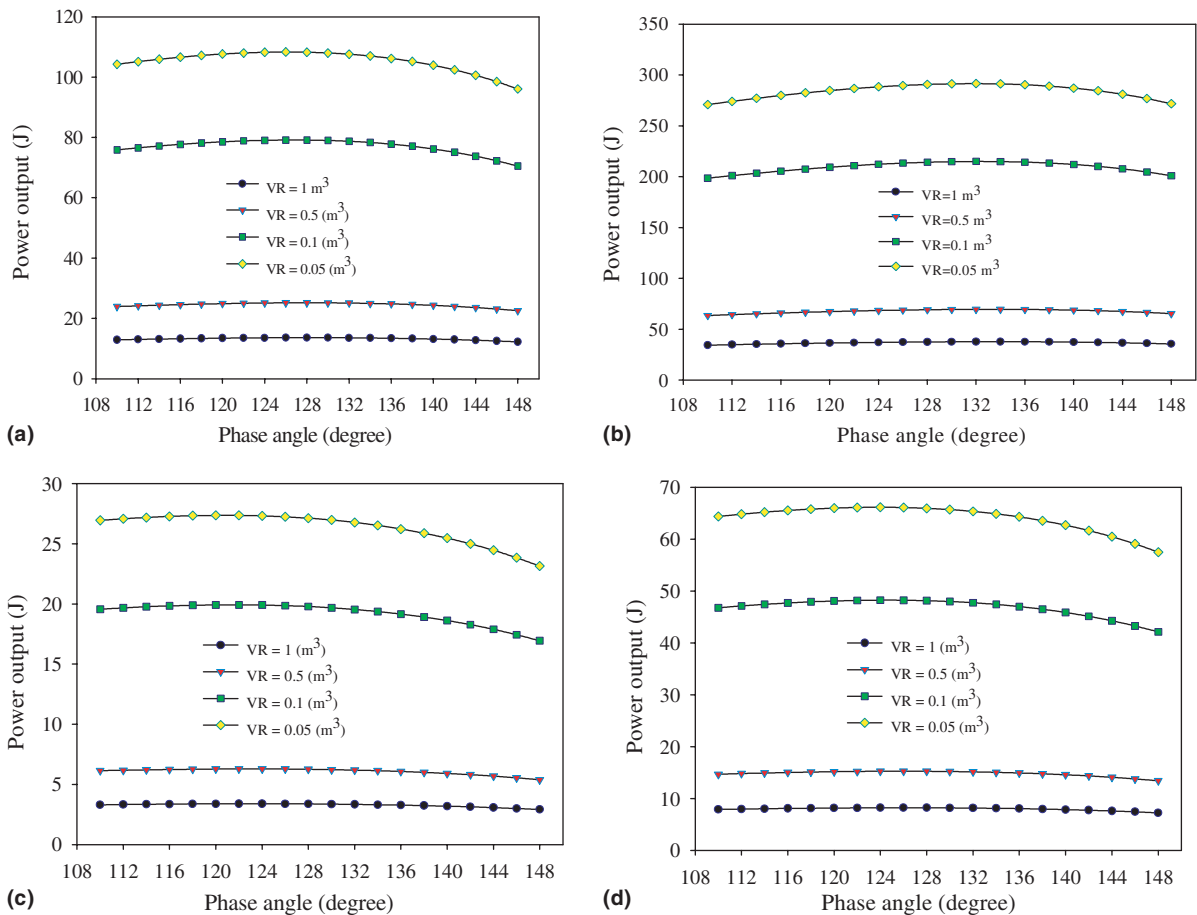


Fig. 9. Simulated power outputs at varying phase angle and different T_E : (a) 100°C ; (b) 200°C ; (c) 50°C ; (d) 75°C .

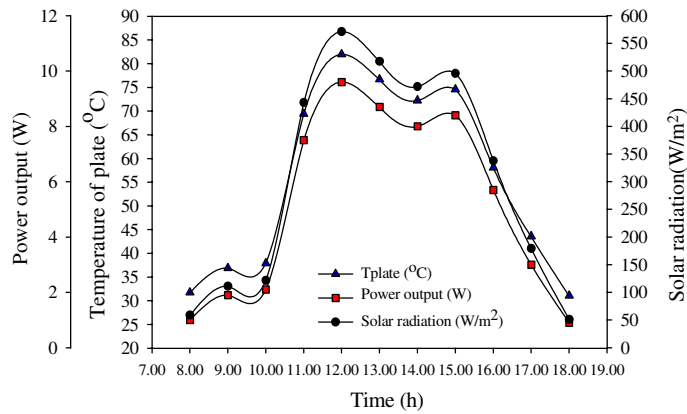


Fig. 10. The relationship between solar radiation and simulated power output ($V_R = 0.5\text{ m}^3$).

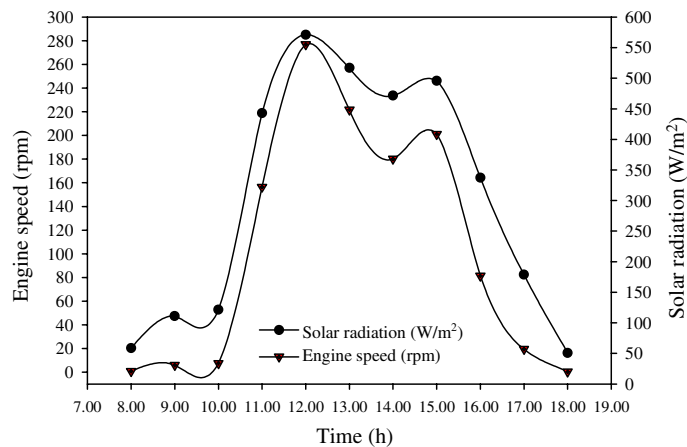


Fig. 11. Relation between solar radiation and engine speed ($V_R = 0.5\text{ m}^3$).

the day. A compromise must be sought which could be used throughout the day. V_R could be chosen so that error in total mass will be at an acceptable minimum. Fig. 9 shows simulated power output graphs. The graphs show that output power is a function of hot-side temperature and V_R . Power output is low at high V_R , but it is high at low V_R . Fortunately, the effect of V_R on power output is low if V_R is greater than 0.5 m^3 , but when V_R is equal to 1, output is only half of that when V_R is 0.5 . Therefore, $V_R = 0.5\text{ m}^3$ is suitable for this simulation. Then, the compromised choice of the phase angle would be around 140° with the associated error in mass of 1.2% in the cyclical process.

Fig. 10 shows the relationship between solar radiation and simulated power output and efficiency of solar thermal engine. It was the result of calculation by using the solar radiation data and estimated temperatures (T_E) and (T_C). Temperature (T_E) was estimated as 75% of

(T_{plate}) and temperature ($T_C = 30$) was taken approximately as 75% of water temperature.

Fig. 11 shows the relationship between solar radiation and the rpm available from the solar engine considered. The maximum rpm achievable was 120. Since solar radiation is low during morning hours and evening hours, the speed of the solar engine would be reduced accordingly.

6. Conclusion

Although, many new designs have been introduced recently and the developments in this field need to be updated, a new solar thermal engine for circulating water of canals, lakes, fish and shrimp ponds, and irrigation reservoirs was studied numerically. The power output versus the energy output characteristic, heat conductance

characteristic from solar radiation to conceivable heat on the collector box, and imperfect regeneration coefficient characteristic were simulated. Results showed that the power and consequently the crankshaft velocity of this system depend highly on the thermal performance of the solar collector. To meet the assumption of constancy of mass in the system, regenerator volume must be carefully chosen, and the phase angle must be set at a value to give maximum power output for the whole day. Based on simulation results, an experimental prototype of this engine is under deployment, and results will be published in a coming paper. This system once validated with experiments, is expected to help reducing stagnant water pollution in both rural and urban areas and to promote the use of solar energy as well.

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